TITLE OF THE INVENTION

PISTON COMPRESSOR

BACKGROUND OF THE INVENTION

The present invention relates to a piston compressor for a vehicular air conditioner and, more particularly, to a technology for restraining deformation of a cylinder block.

For example, Japanese Laid-Open Patent Publication No. 8-14160 discloses a gasket 101 as shown in Fig. 13. The gasket 101 is used for a piston compressor for a vehicular air conditioner.

The gasket 101 is formed with a plurality of through holes 103 that substantially coincide with opening edges of cylinder bores 102 each containing a piston, a plurality of insertion holes 105 through which through bolts 104 are inserted, and a center hole 106 through which a drive shaft is inserted. As a piston compressor provided with this gasket 101, a piston compressor is known in which as shown in a partially enlarged cross-sectional view of Fig. 14, a front housing member 108 is joined to a front end face (left-hand side in the figure) of a cylinder block 107, a rear housing member 110 is joined to a rear end face (right-hand side in the figure) thereof via a valve plate 109, and these three elements are fastened to each other by the through bolts 104. In this piston compressor, the gasket 101 is interposed between the cylinder block 107 and the valve plate 109. As shown in Fig. 15, the cylinder block 107 is formed with the cylinder bores 102 and an accommodation chamber 111 for accommodating a rotary valve for sucking refrigerant gas.

In the piston compressor described in the above-described

Publication, when the through bolts 104 are tightened, the cylinder block 107 is subjected to bending moment and is thus deformed. Specifically, as shown in Fig. 14, in the state in which the through bolts 104 are tightened, on a joint surface between the cylinder block 107 and the front housing member 108, a specific pressure f1 acts on the front end face of the cylinder block 107 from the front housing member 108. Also, on a joint surface between the cylinder block 107 and a seal surface of the gasket 101, a specific pressure f2 acts on the rear end face of the cylinder block 107 from the valve plate 109.

Taking one arbitrary point on the front end face of the cylinder block 107, on which the specific pressure f1 acts, as action point P1, and taking one arbitrary point on the rear end face of the cylinder block 107, on which the specific pressure f2 acts, as action point P2, bending moment M acts around the center P3 of straight line H connecting P1 and P2. By this bending moment M, a force Fm in a radial direction of the gasket 101 is applied to both of the action points P1 and P2, by which the cylinder block 107 is deformed as indicated by two-dot chain lines shown in Fig. 15. As a result, there is a fear that smooth reciprocating motion of the piston is hindered by this deformation.

Also, in a case where the accommodation chamber 111 for the rotary valve is formed in the cylinder block 107 as shown in Fig. 15, the accommodation chamber 111 is easily deformed because the rigidity of the cylinder block 107 is low. Therefore, smooth rotation of the rotary valve can be hindered.

SUMMARY OF THE INVENTION

An object of the present invention is to provide a piston compressor in which bending moment acting on a cylinder block

is reduced to restrain deformation of the cylinder block, and the motion of a piston and a rotary valve is performed smoothly to enhance the durability of the piston compressor.

To achieve the foregoing and other objectives and in accordance with the purpose of the present invention, a piston compressor having a cylinder block, a front housing member, a rear housing member, a through bolt, a plurality of pistons, a drive shaft, and a gasket is provided. The cylinder block has a plurality of cylinder bores. The cylinder block has two end faces at which the cylinder bores open. The front housing member is secured to one of the end faces of the cylinder block. The rear housing member is secured to the other one of the end faces of the cylinder block with a valve plate in between. The through bolt fastens the cylinder block, the rear housing member, and the front housing. Each piston is accommodated and reciprocates in one of the cylinder bores. The drive shaft drives the pistons, and is rotatably supported by the cylinder block. Reciprocation of the pistons compress and discharge refrigerant gas. The gasket is located between the cylinder block and the valve plate. The gasket has a center hole and a plurality of bore holes. Each bore hole is aligned with one of the cylinder bores. A first through hole is formed in the gasket to reduce bending moment generated in the cylinder block when the through bolt is fastened. The first through hole is located between an adjacent pair of the bore holes and in an imaginary circle. The center of the imaginary circle coincides with the center of the bore hole, and the radius of the imaginary circle is a first radius. The first radius is the distance from the center of the gasket to the center of one of the bore holes.

Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example

the principles of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

Fig. 1 is a cross-sectional view of a piston compressor in accordance with a first embodiment of the present invention;

Fig. 2 is a partially cross-sectional view of the compressor shown in Fig. 1;

Fig. 3 is a front view of a gasket provided in the compressor shown in Fig. 1;

Fig. 4 is a front view of a conventional gasket used for explanation of the first embodiment;

Fig. 5 is a graph showing a relationship between circumferential lengths of seal portions necessary for function and distances from the gasket center in a gasket;

Fig. 6 is a graph showing a relationship between circumferential lengths of seal portions unnecessary for function and distances from the gasket center in a gasket;

Fig. 7 is a graph showing a total change amount of bending moment generated in a cylinder block;

Fig. 8 is a cross-sectional view of a piston compressor in accordance with a second embodiment;

Fig. 9 is a front view of a gasket provided in the compressor shown in Fig. 8;

Fig. 10 is a front view of a conventional gasket used for explanation of a second embodiment;

Fig. 11 is a front view of a gasket in a modified embodiment;

Fig. 12 is a front view of a gasket in another modified

embodiment;

Fig. 13 is a front view of a prior art gasket;

Fig. 14 is a partially cross-sectional view of a prior art piston compressor; and

Fig. 15 is a partially cross-sectional view of a prior art piston compressor.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

A first embodiment of the present invention will now be described in detail with reference to Figs. 1 to 7.

In a variable displacement piston compressor in accordance with the first embodiment of the present invention, as shown in Fig. 1, a front housing member 3 is joined to a front end face of a cylinder block 1 via a gasket 2, and a crank chamber 4 serving as a control chamber is defined on the inside thereof. Also, a rear housing member 6 is joined to a rear end face of the cylinder block 1 via a valve plate 5, and a discharge chamber 7 and a suction chamber 8 are defined on the inside thereof. Between the cylinder block 1 and the valve plate 5 is interposed a gasket 9, and between the valve plate 5 and the rear housing member 6 are interposed a discharge valve forming plate 10 formed integrally with a discharge valve and a retainer forming plate 11 for forming a retainer. The cylinder block 1, the front housing member 3, and the rear housing member 6 are fastened by through bolts 12, not shown in Fig. 1.

In shaft holes formed in central portions of the cylinder block 1 and the front housing member 3, a drive shaft 13 is rotatably supported by radial bearings 14a and 14b. In a front end portion of the drive shaft 13 is provided a shaft seal device 15. In the crank chamber 4, a lug plate 16 is fixed to the drive shaft 13 so as to be integrally rotatable,

and a swash plate 17 serving as a cam plate is disposed in a state in which the drive shaft 13 is inserted through a through hole formed in the swash plate 17. A hinge mechanism 18 is interposed between the lug plate 16 and the swash plate 17. The swash plate 17 can be rotated in synchronism with the lug plate 16 and the drive shaft 13 by a hinge connection between the swash plate 17 and the lug plate 16 via the hinge mechanism 18 and the support of the drive shaft 13, and also can be tilted with respect to the drive shaft 13 while sliding in an axial direction of the drive shaft 13.

A plurality of cylinder bores 19 arranged in a circumferential direction in the cylinder block 1 each contain a piston 20 capable of reciprocating. Between each piston 20 and the valve plate 5, a compression chamber 21 whose volume is changed according to reciprocating motion of the piston 20 is defined. Each piston 20 is engaged with a peripheral edge portion of the swash plate 17 via a pair of shoe 22. Therefore, rotational motion of the swash plate 17 performed via the lug plate 16 and the hinge mechanism 18, which is caused by rotation of the drive shaft 13, is converted to reciprocating motion of the pistons 20 performed via the shoes 22. The lug plate 16, the swash plate 17, the hinge mechanism 18, and the shoes 22 constitute a crank mechanism that converts the rotational motion of the drive shaft 13 to compressive motion for compressing refrigerant gas in the compression chamber 21.

A rotary valve accommodating chamber 23 is formed in the cylinder block 1, and in the rotary valve accommodating chamber 23, a rotary valve 24 is connected to the drive shaft 13 via a coupling 25 so as to be rotatable in synchronism with the drive shaft 13. In the rotary valve 24, a suction passage 26 that always communicates with the suction chamber 8 is formed, and an outlet 27 of the suction passage 26 is open in

an outer peripheral surface of the rotary valve 24. In the cylinder block 1, communication holes 28 are formed. Each communication hole corresponds to one of the compression chambers 21 and allows the outlet 27 of the rotary valve 24 to communicate with the corresponding compression chamber 21.

When the drive shaft 13 of the compressor is rotated by engine power, the swash plate 17 is rotated via the lug plate 16 and the hinge mechanism 18, so that the pistons 20 are reciprocated in the cylinder bores 19 via the shoes 22. On a suction stroke of the piston 20, the outlet 27 of the rotary valve 24 is connected to each communication hole 28, so that the refrigerant gas in the suction chamber 8 is sucked into each compression chamber 21 through the suction passage 26. Further, when each piston 20 takes compression stroke and discharge strokes, the corresponding communication hole 28 is closed by an outer peripheral surface of the rotary valve 24, so that the refrigerant gas in the compression chamber 21 pushes away the discharge valve and is discharged to the discharge chamber 7.

Next, an essential point of the present invention will be described in detail. First, forces acting on the cylinder block 1 in this embodiment are shown in Fig. 2. In a state in which the through bolts 12 are tightened, on a joint surface between the cylinder block 1 and the front housing member 3, a specific pressure f1 acts on a front end face of the cylinder block 1 from the front housing member 3. Also, on a joint surface between the cylinder block 1 and a seal surface of the gasket 9, a specific pressure f2 acts on a rear end face of the cylinder block 1 from the gasket 9.

Taking one arbitrary point on the front end face of the cylinder block 1, on which the specific pressure f1 acts, as action point P1, and taking one arbitrary point on the rear

end face of the cylinder block 1, on which the specific pressure f2 acts, as action point P2, bending moment M acts around the center P3 of straight line H connecting P1 and P2. When the shortest distance between both of the action points P1 and P2 in a radial direction of the gasket 9 is taken as D1, the shortest distance therebetween in the axial direction of the through bolt 12 is taken as D2, and a radial force generated at both of the action points P1 and P2 by the bending moment M is taken as Fm, the bending moment M is obtained by the following formulae:

$$Fm = f2 \cdot (D1/D2) \quad \ldots (1)$$

$$M = Fm \cdot D2 = f2 \cdot D1 \quad \ldots (2)$$

From these two formulae, it is found that the force Fm and the bending moment M increase as the specific pressure f2 acting on the rear end face of the cylinder block 1 from the gasket 9 increases, or as the action point P2 is closer to the center of the gasket 9.

The gasket 9 in this embodiment is shown in Fig. 3. The gasket 9 is formed of a rigid base consisting of an iron-base metallic sheet and an elastic layer having sealing ability, such as rubber, with which both surfaces of the base are coated. Also, the gasket 9 has a plurality of (six in this embodiment) bore holes 29 that substantially coincide with the opening edges of the cylinder bores 19 and a plurality of (six in this embodiment) bolt holes 30 through which the through bolts 12 are inserted. In a circle whose radius is a distance Rb from the center of the gasket 9 to the center of each bore hole 29, a through hole is formed which corresponds to a center hole 31 (in a circle indicated by dotted line in Fig. 3) in the conventional gasket and first through holes 32 communicating with each other. Between a circle having a radius of a distance Rb from the center of the gasket 9 and a

circle having a radius Rc from the center of the gasket 9, second through holes 33 are formed. As is apparent from Fig. 2, in the range in which the first through holes 32 and the second through holes 33 are provided, the specific pressure f2 does not act on the cylinder block 1, so that bending moment is not generated. Because the bending moment is larger at a position closer to the center of the gasket 9, the provision of the through holes 32 and 33 can reduce the bending moment.

The meaning of the radius Rc and a method for determining the same will be explained with reference to Figs. 4 to 7. Fig. 4 shows a conventional gasket 34 formed with bore holes 29, bolt holes 30, and a center hole 31. In Fig. 4, solid line hatched portions are seal portions that are necessary for function of sealing the bore holes 29, the bolt holes 30, and the interior of the compressor. That is to say, in the gasket 34, the range excluding the solid line hatched portions, the bore holes 29, the bolt holes 30, and the center hole 31 (dotted line hatched portions in Fig. 4) indicates portions that are unnecessary for the function of the gasket. The length of seal portions that are necessary for the function on the circumference of a circle whose radius is a certain distance x from the center O of the gasket 34 and the length of seal portions that are unnecessary for the function on the circumference of a circle whose radius is a certain distance x from the center O are represented by graphs of Figs. 5 and 6, respectively. Rg indicates the radius of the gasket 34. Here, a complement is given to the description of "the length of seal portions on the circumference of a circle whose radius is a certain distance x from the center O of the gasket 34". For example, when the length of seal portions that are necessary for the function on the circumference of a circle whose radius is a distance A from the center O is taken as La, and the length of seal portions that are unnecessary for the function thereon is taken as Lb, as is apparent from Fig. 4, La and Lb

are expressed as

$$La = L1 + L3 + L5 + L7 + L9 + L11$$
$$Lb = L2 + L4 + L6 + L8 + L10 + L12$$

From Figs. 5 and 6, an area S of seal portions of the gasket 34 is calculated by the following formula (3).

$$S = \int_0^{Rg} f(x)dx + \int_0^{Rb} g(x)dx + \int_{Rb}^{Rg} h(x)dx \qquad (3)$$

In the above formula (3), the function f(x) is a function for the graph of Fig. 5, the function g(x) is a function for the range of $0 \le x \le Rb$ in the graph of Fig. 6, and the function h(x) is a function for the range of $Rb \le x \le Rg$ in the graph of Fig. 6.

Further, when the total pressure applied to the whole of a seal surface of the gasket 34 at the time of tightening of the through bolts 12 is taken as F, the specific pressure f2 per unit area of the seal surface is expressed as

$$f2 = F/S$$

The total pressure F depends on the tightening force of bolt, and the shape, rigidity of the cylinder block and rear housing member, and it is thought that the total pressure F in this embodiment is equivalent to that of the conventional compressor.

Next, it is assumed that through holes with a minute width Δx are provided in the portions that are unnecessary for the function (dotted line hatched portions in Fig. 4) on the circumference whose radius is a certain distance x from the center O. An area S(x) of seal portions at this time is calculated by the following two formulae.

$$\begin{cases} S(x) = S - \int_x^{x+\Delta x} g(x)dx & \text{(when } 0 \leq x \leq Rb) \quad (4) \\ S(x) = S - \int_x^{x+\Delta x} h(x)dx & \text{(when } Rb \leq x \leq Rg) \quad (5) \end{cases}$$

When the increase in specific pressure at the time when the through holes with a minute width Δx are provided is taken as $\Delta f2$, $\Delta f2$ can be expressed as $\Delta f2 = F/S(x) - F/S$ using the above-described formulae (4) and (5).

Therefore, taking the increase in bending moment as $\Delta M1$, $\Delta M1$ can be expressed by the following formula (6) using the above-described formula (1) and the above-described $\Delta f2$.

$$\Delta M1 = \int_0^{Rg} (\Delta f2 \cdot x)dx \qquad (6)$$

Also, taking the decrease in bending moment due to the provision of through holes as $\Delta M2$, from the above-described formula (2), $\Delta M2$ is expressed as

$$\Delta M2 = f2 \cdot x \qquad \ldots (7)$$

Therefore, when the total change amount of bending moment at the time when the through holes are provided in the portions that are unnecessary for the function on the circumference whose radius is a certain distance x from the center O is taken as ΔM ($= \Delta M2 - \Delta M1$), ΔM is expressed by a graph shown in Fig. 7 using Formulae (6) and (7). Rc is defined as a distance of a point at which $\Delta M1 = \Delta M2$ ($\neq 0$) from the center O. In Fig. 7, Rc denotes a point at which $\Delta M = 0$ (excluding a case where $\Delta M1 = \Delta M2 = 0$ is satisfied).

Fig. 7 means that if through holes are formed in a circle with the radius Rc from the center O, since the decrease in

bending moment due to the through holes is larger than the increase in bending moment due to increased specific pressure, the total bending moment can be decreased.

In this embodiment, a seal portion for sealing the compressor internally and externally is provided in an outer peripheral portion of the gasket 9. As is apparent from Fig. 2, bending moment is not generated on a joint surface 35 between the cylinder block 1 and the gasket 9, which faces a joint surface between the cylinder block 1 and the front housing member 3 in the axial direction of the drive shaft 13. Therefore, it is desirable that the gasket 9 be formed with a seal surface in the range of the joint surface 35 so as to decrease the specific pressure Δf2 as much as possible.

By this embodiment, the bending moment acting on the cylinder block 1 is reduced, and hence the deformation of the cylinder block 1 is restrained. As a result, the deformation of the cylinder bore 19 is restrained, and hence the reciprocating motion of the piston 20 is made smooth. Also, the deformation of the rotary valve accommodating chamber 23 for the rotary valve 24 is restrained, and hence the rotational motion of the rotary valve 24 is made smooth. Further, the specific pressure of gasket is increased by reducing the seal surface, so that the sealing ability of gasket is improved, or sufficient sealing ability of gasket is secured even if the tightening force of bolts is decreased as compared with the conventional compressor. Therefore, the deformation of the cylinder block 1 can further be restrained by the decrease in bolt tightening force, and hence the durability of compressor is enhanced.

Next, a second embodiment will be described with reference to Figs. 8 to 10. In the second embodiment, only points different from the first embodiment shown in Figs. 1 to

7 will be explained. Also, the same reference numerals will be applied to the same or equivalent elements, and the explanation of the elements will be omitted.

Fig. 8 shows a five-cylinder compressor. In this compressor, the rotary valve 24 and the rotary valve accommodating chamber 23 are not used as a suction structure for refrigerant gas, and instead a suction valve forming plate 36 is interposed between the cylinder block 1 and the valve plate 5, and a gasket 37 is interposed between the suction valve forming plate 36 and the cylinder block 1. On the suction stroke of each piston 20, a corresponding suction valve is opened, and a refrigerant gas passes through a corresponding suction hole formed in the valve plate 5 and is sucked into the compression chamber 21. Further, when the piston 20 takes compression and discharge strokes, the suction valve is closed, and the suction hole is closed and the refrigerant gas in the compression chamber 21 pushes away the discharge valve and is discharged to the discharge chamber 7.

As shown in Fig. 9, in the gasket 37 used in this embodiment, one through hole 38 is formed in a state in which the center hole 31 (in a circle indicated by dotted line in Fig. 9), the first through holes, and the second through holes communicate with each other. In the piston compressor of this embodiment, the number of cylinders is decreased to five as compared with the above-described first embodiment. Fig. 10 shows a conventional gasket 39 used for a five-cylinder piston compressor. In Fig. 10, hatched portions are seal portions that are necessary for function of sealing the bore holes 29, the bolt holes 30, and the interior of the compressor. As is apparent from Fig. 10, in the gasket 39, seal portions that are unnecessary for the function are present even between the adjacent bore holes 29. Therefore, as in the gasket 37 of this embodiment, it is possible to form the integral through

hole 38 by allowing the center hole 31, the first through holes, and the second through holes to communicate with each other. Thereby, the bending moment is reduced, and resultantly the deformation of the cylinder block 1 is restrained. Also, by forming the integral through hole 38 in this manner, a mold necessary for manufacturing the gasket 37 is formed easily, and the life of mold is extended, which also achieves an effect of reducing the manufacturing cost.

It should be apparent to those skilled in the art that the present invention may be embodied in many other specific forms without departing from the spirit or scope of the invention. Particularly, it should be understood that the invention may be embodied in the following forms.

As shown in Figs. 11 and 12, the center hole 31 and the first through holes 32 may be separated from each other.

In these examples as well, the deformation of the cylinder block is restrained by reducing bending moment, and hence the motion of the piston and rotary valve is made smooth, by which the durability of the piston compressor is enhanced.

The present examples and embodiments are to be considered as illustrative and not restrictive and the invention is not to be limited to the details given herein, but may be modified within the scope and equivalence of the appended claims.